

Leaner Lifted-Flame Combustion Enabled by the Use of an Oxygenated Fuel in an Optical CI Engine

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ABSTRACT

Leaner lifted-flame combustion (LLFC) is a mixing-controlled combustion strategy for compression-ignition (CI) engines that does not produce soot because the equivalence ratio at the lift-off length is less than or equal to approximately two. In addition to completely preventing soot formation, LLFC can simultaneously control emissions of nitrogen oxides because it is tolerant to the use of exhaust-gas recirculation for lowering in-cylinder temperatures. Experiments were conducted in a heavy-duty CI engine that has been modified to provide optical access to the combustion chamber, to study whether LLFC is facilitated by an oxygenated fuel blend (T50) comprising a 1:1 mixture by volume of tri-propylene glycol mono-methyl ether with an ultra-low-sulfur #2 diesel emissions-certification fuel (CFA). Results from the T50 experiments are compared against baseline results using the CFA fuel without the oxygenate. Experimental measurements include crank-angle-resolved natural luminosity and chemiluminescence imaging. Dilution effects were studied by adding nitrogen and carbon dioxide to the intake charge. Initial experiments with a 2-hole fuel-injector tip achieved LLFC at low loads with the T50 fuel, and elucidated the most important operating parameters necessary to achieve LLFC. The strategy was then extended to more moderate loads by employing a 6-hole injector tip, where lowering the intake-manifold temperature, reducing the coolant temperature, and retarding the start-of-combustion timing resulted in sustained LLFC at both 21% and 16% intake-oxygen mole fractions at loads greater than 5 bar gross indicated mean effective pressure. In contrast to the results with T50, LLFC was not achieved under any of the test conditions with CFA.

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INTRODUCTION

Direct-injection CI engines can achieve high efficiencies and high loads, but they require expensive aftertreatment systems to comply with ever-more-stringent emissions regulations. Tradeoffs between emissions of nitrogen oxides (NO_x) and smoke (or particulate matter, PM) have forced vehicle manufacturers to employ combinations of aftertreatment systems to achieve legislated emissions targets with as little impact on fuel economy as possible. Current aftertreatment options for diesel vehicles include: diesel particulate filters to capture PM, selective catalytic reduction or lean- NO_x trap systems to control NO_x , and diesel oxidation catalysts to oxidize hydrocarbon (HC) and carbon monoxide (CO) emissions [1,2,3]. Although aftertreatment systems can be very effective, they can also be expensive, large, heavy, and require continual management as well as an additional on-board fluid. Hence, there is interest in developing combustion strategies that reduce the amount of aftertreatment required to meet legislated emissions targets.

Advanced combustion strategies such as globally premixed, low-temperature combustion (LTC) can lower both NO_x and PM emissions [4,5,6,7,8,9,10,11], but they are less suitable for high-load conditions than mixing-controlled combustion due to the de-coupling

of the fuel-injection and ignition processes (i.e., ignition timing is controlled by chemical kinetics rather than injection timing). This decoupling can lead to control, noise, and emissions issues.

Leaner lifted-flame combustion (LLFC) is an alternative advanced-combustion strategy wherein ignition timing is still controlled by injection timing. LLFC is a form of mixing-controlled diesel combustion that can eliminate soot if the equivalence ratio at the flame lift-off location (defined as the axial distance between the fuel-injector orifice exit and the position where the standing premixed autoignition zone stabilizes during mixing-controlled combustion [12,13]) is maintained at values less than two [14,15]. Operating in this mode may potentially allow meeting even the strictest soot particle number regulations without the use of a diesel particulate filter, while simultaneously avoiding many of the control, noise, and emissions problems associated with globally premixed LTC strategies.

Early investigations of the applicability of LLFC in a heavy-duty diesel engine showed that LLFC could be sustained by using moderate to high dilution levels and a two-hole injector tip [15]. For higher-flow injectors (6- and 10-hole), LLFC could not be sustained

due to shortening of the lift-off length (H) by re-entrainment of hot combustion products upstream of H, and by interactions between adjacent jets as injector tips with more orifices are used.

One method that has shown promise to extend the possible engine operating range for LLFC is to use fuels with molecular compositions and structures that have a lower tendency to form soot, such as oxygenates [16]. An oxygenate is a compound that has oxygen chemically bound within the fuel molecule. Classes of oxygenates include esters, ethers, alcohols, and ketones. For the same amount of charge-gas oxygen entrained upstream of H, the presence of oxygen in the fuel molecule would bring the fuel-air mixture closer to stoichiometric conditions (i.e., it would be leaner) relative to the mixture that would be produced if a non-oxygenated fuel was used, hence the lower smoke emissions [17]. A comprehensive research effort beginning with approximately 71 potential oxygenated candidates was narrowed down to a single compound that showed the most promise for blending with diesel fuel, tripropylene-glycol monomethyl ether (TPGME, $C_{10}H_{22}O_4$). This selection was made on the basis of a number of factors including total PM emissions, gaseous emissions, flash point, toxicity, and other practical considerations [18,19,20]. Further, the most effective chemical reactions for preventing soot formation are those where each oxygen (O) atom from the fuel is bonded to a different carbon (C) atom in the products, reducing the pool of C atoms available for soot formation [20]. This explains the selection of TPGME over di-butyl maleate (DBM), for example. DBM has two $---O---(C=O)---$ ester groups in its molecular structure, and these ester groups can directly lead to the production of carbon dioxide (CO_2), where two O atoms from the fuel are bonded to the same C atom in the products. In contrast, the distribution of O atoms in the TPGME molecule ensures that each O atom reacts to form a CO molecule. These C=O bonds do not break at combustion temperatures [21], thereby preventing a greater fraction of the C atoms in the fuel from contributing to the pool of available soot-precursor species.

The objective of the current study was to determine if a 50/50 blend by volume of TPGME with an ultra-low-sulfur #2 diesel baseline fuel (the blend T50) could enable LLFC at less extreme conditions than the neat baseline fuel. In the first part of the study, a 2-hole injector tip was used, and the goals included identifying the extent to which T50 could enable sustained LLFC by lowering the equivalence ratio at H in the absence of re-entrainment and interactions between adjacent jets [22]. Experiments were conducted at three fuel-injection pressures with three levels of charge-gas dilution. Diagnostics included conventional heat-release analysis, engine-out emissions, and high-speed in-cylinder imaging of both natural luminosity and electronically excited hydroxyl radical (OH^*) chemiluminescence. This study was used to determine the conditions and operating parameters that would have the most significant effect on the measured equivalence ratio at H, and thus determine which conditions would best lead to LLFC. The most significant and promising parameters were then adjusted toward maximum effect in the second part of the study, in which a 6-hole injector tip was used to investigate the possibility of achieving LLFC at higher engine loads using the T50 fuel.

Although the basic concept of LLFC was first demonstrated in constant-volume combustion-vessel experiments over a decade ago [14], its successful implementation in engines has proven elusive. Previous engine studies have demonstrated mixing-controlled combustion with low or undetectable engine-out soot emissions, but this has typically been achieved by oxidizing the soot that has been produced before the exhaust valves open. The current engine study is different, because it demonstrates LLFC, which *prevents soot formation* and hence does not rely on soot oxidation before the exhaust valves open. Other previous engine-research efforts have attenuated soot emissions by employing massive levels of intake-charge dilution to lower in-cylinder temperatures sufficiently to prevent soot formation (e.g., [23]) or by using neat oxygenates as the fuel to attenuate soot formation and/or enhance soot oxidation (e.g., [24,25]). The current study is different, because it does not rely on high dilution, and it does not use a neat oxygenate as the fuel (though it does employ fuel oxygenation to facilitate the achievement of LLFC).

To the authors' knowledge, the only previous attempts similar to the approach employed herein are described in the papers by Chartier et al. [26] and Polonowski et al. [15]. Using a 3-hole injector tip, Chartier et al. [26] were able to drive engine-out smoke emissions to zero and approach LLFC. They did not achieve LLFC, however, as evidenced by their observations that "line-of-sight natural luminescence images of the whole combustion chamber showed that soot was present in the downstream jet region" and this "is further supported by the fact that the glass piston had to be cleaned regularly because of significant soot deposits on its surface" [26]. Polonowski et al. [15] also approached LLFC, but were unable to sustain it with a 6-hole injector tip using #2 diesel fuel. Hence, to the authors' knowledge, the present study is the first to conclusively demonstrate sustained LLFC in an engine at moderate load using a realistic injector configuration (i.e., a 6-hole tip) and a feasible fuel.

EXPERIMENTAL SETUP

Engine and Fuel System

The experiments were performed in a single-cylinder version of a heavy-duty engine modified to provide optical access to the combustion chamber. A schematic and the primary specifications of this optical engine are provided in [Figure 1](#) and [Table 1](#), respectively.

The major differences between the optical engine and a comparable production engine relate to the piston bowl, which has a flat bottom, vertical walls, and a larger land above the top compression ring. The larger top-ring land can allow optical access near top dead center (TDC) through a window in the cylinder wall and a notch in the piston bowl-rim, and it prevents the rings from riding over the cylinder-wall windows. The 2-hole experiments used such a notched bowl-rim, but the 6-hole experiments used a solid bowl-rim without the side optical access in order to avoid jet interaction with the notch. The differences between bowl rims led to a lower geometric compression ratio (CR) of 12.2:1 for the 2-hole configuration used, and 12.7:1 for the 6-hole configuration. A view of the differences in geometry is presented in [Figure 2](#).

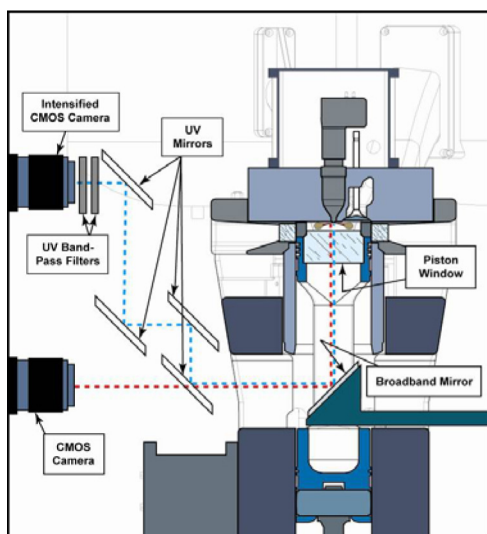


Figure 1. Optical engine schematic. Top and bottom cameras are used for chemiluminescence and natural luminosity imaging, respectively.

Table 1. Optical Engine Specifications

Research engine	Single-cylinder
Cycle	4-stroke CIDI
Valves per cylinder	4
Bore	125 mm
Stroke	140 mm
Displacement per cylinder	1.72 liters
Intake valve open ^a	32° BTDC exhaust
Intake valve close ^a	153° BTDC compression
Exhaust valve open ^a	116° ATDC compression
Exhaust valve close ^a	11° ATDC exhaust
Connecting rod length	225 mm
Connecting rod offset	None
Piston bowl diameter	90 mm
Piston bowl depth	16.4 mm
Squish height	1.5 mm
Swirl ratio ^b	0.59
Compression ratio (geometric)	12.7:1 (6-hole setup), 12.2:1 (2-hole setup)
Compression ratio (effective)	11.1:1 (6-hole setup), 11.0:1 (2-hole setup)
Simulated compression ratio	16.0:1

^aAll valve timings are for 0.03 mm lift

^bMeasured at the Caterpillar Technical Center using an AVL swirl meter

In both cases, the CR of the optical engine is lower than the ~16:1 CR for a comparable production engine. To compensate for the lower CR, the intake pressure is elevated to closely match the conditions in the production engine over the crank-angle range during which combustion occurs [27].

The optical engine was skip-fired with only one injection event every 5 engine cycles for the two-hole configuration, and one injection event every 15 engine cycles for the 6-hole configuration. This approach was used to lower the magnitude of the thermal transient that results from switching between motored and fired operation, and the risk of window failure due to the corresponding thermal and mechanical stresses [15]. The different skip-firing frequencies were designed to keep the fuel flow rate and concentration of soot in the exhaust stream approximately equal in the 2-hole and 6-hole tip experiments, to facilitate smoke meter measurements at the low engine-out smoke levels of this study.



a. notched bowl-rim



b. solid bowl-rim

Figure 2. Renderings of piston bowl-rim configurations. The two-hole experiments had a notched bowl-rim configuration (a), while the six-hole experiments used a solid bowl-rim (b). This led to a difference in geometric compression ratio of 0.5 between these experiments.

The residual exhaust-gas that remains trapped in the combustion chamber after a fired cycle is replaced with fresh intake charge (i.e., air, N₂, and CO₂) during the subsequent motored cycles. The heat rejection from the combustion chamber to the coolant is lower due to the skip-fired operation and short run-times, which requires the optical engine to use a heater to maintain a constant coolant temperature.

Fuel delivery was accomplished using a high-pressure common-rail fuel-injection system, the principal components of which include a diaphragm pump (Newport Scientific, Model 46-16060) and a near-production Caterpillar electronically actuated solenoid-operated fuel injector. The fuel-wetted internals of the pump are fabricated from stainless steel, Kalrez, or Teflon, making the system suitable for experiments with a wide range of conventional and unconventional fuels having high corrosivities, low viscosities, or other properties that can cause conventional piston-type pumps to fail.

Fuel-injector specifications are provided in Table 2. A commercial X-ray tomography technique (as described in [28]) was used to confirm the orifice diameter, length-to-diameter ratio, and included spray angle listed in Table 2. Figure 3 presents typical injection-rate profiles for the 6-hole tip at 240 MPa, as measured at the outlet of one injector orifice using a momentum-based rate-of-injection meter developed at Sandia [29].

Table 2. Common-rail fuel-injector specifications.

Injector type	Caterpillar CR350
Injector tip style	Mini-sac
Number of injector orifices	2 or 6
Orifice diameter	110 μm
Orifice length-to-diameter ratio	8
Hydro-erosion	20 $\pm$ 2%
Included spray angle	140°

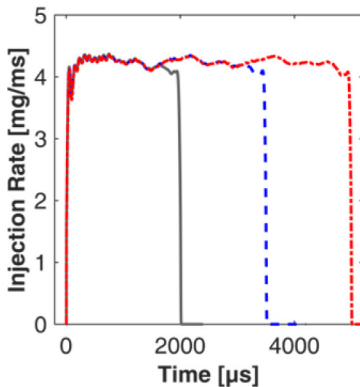


Figure 3. Typical injection-rate profiles for the 6-hole tip at three different injection durations and a single injection pressure (240 MPa).

A flush-mounted, water-cooled pressure transducer (AVL Model QC32C) was used to measure cylinder-pressure every 0.5 CAD. Gas analyzers (California Analytical Instruments, Series 600) measured CO, CO₂, O₂, NO_x, and HC emissions. Engine-out soot emissions were measured using both a smoke meter (AVL Model 415S) and a prototype laser-induced incandescence instrument (see Figure 5 of [30] and [24]). Reference [31] describes in detail the procedures used for cylinder-pressure data acquisition and how the skip-fired exhaust-gas measurements were converted to values similar to those that would be obtained if the engine was operated in a continuous-fired mode. References [15,27] provide further details of the experimental setup and procedures.

Fuels

The T50 test fuel was a 50/50 blend by volume of TPGME (see Figure 4) and the ultra-low-sulfur #2 diesel emissions-certification fuel, batch A, used in previous studies [15,32,33,34], and denoted as CFA in the present paper. TPGME has an oxygen content of 31% by mass and is produced commercially as a mixture of up to eight isomers [35] by reacting propylene oxide with methanol [36]. It is used in the manufacture of polyester plastics, as a solvent for household and industrial cleaners; and for ballpoint and felt-tipped pens and inkpads (to prevent drying). CFA was also used as the baseline fuel. Selected properties of CFA, TPGME, and T50 are presented in Table 3, where V_{oxy} is the volume fraction of the oxygenate, A/F_{st} is the stoichiometric air-to-fuel ratio (by mass), Y represents mass fraction, ρ is the density, CN is the derived cetane number (as quantified by D6890), and q_{LHV} is the lower heating value (refer to [15,33,34] for detailed CFA properties). Due to its oxygen content, TPGME has a specific energy (energy per unit mass) that is 35.4% lower than the baseline fuel, but this effect is somewhat mitigated by blending it with CFA.

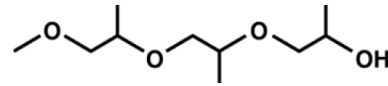
Figure 4. Molecular structure of one isomer of tripropylene glycol monomethyl ether (TPGME, C₁₀H₂₂O₄).

Table 3. Selected properties of the test fuels.

Parameter	CFA	TPGME	T50
V_{oxy} [%]	0.0	100	50.0
A/F_{st}	14.5	13.1	13.8
Ω_f [%]	0.0	12.9	5.79
Y_C [%]	87.0	58.2	71.7
Y_H [%]	13.0	10.7	11.8
Y_O [%]	0.0	31.0	16.5
ρ [kg/m ³]	848	963	906
CN ^a [-]	43.7	81.3	62.5 ^b
q_{LHV} [MJ/kg]	42.9	27.7	34.8

^aDerived cetane number (ASTM D6890)

^bLinear volumetric 50/50 blend model of CFA and TPGME for derived CN.

The oxygen ratio of a fuel (Ω_f) in Table 3 is defined as the amount of oxygen in the fuel mixture divided by the amount of oxygen required to convert all fuel elements to CO₂ and water (H₂O) [17]:

$$\Omega_f = \frac{n_o}{2n_c + \frac{1}{2}n_h} \quad (1)$$

where n_o , n_c , and n_h are the numbers of O, C, and H atoms in the fuel, respectively, and it is assumed that the fuel mixture does not initially contain CO₂ or H₂O.

Optical Diagnostics

Two high-speed cameras were used to simultaneously record natural luminosity (NL) and OH* chemiluminescence. NL refers to the broadband light emitted by the combustion process during a fired cycle and used to indicate the locations of high-temperature soot particles during mixing-controlled combustion [37]. In this study, NL signal in the range of 380-1000 nm was collected every 0.5 crank angle degrees (CAD) using a high-speed CMOS camera (Vision Research, Phantom v7.3) fitted with a 35-mm Nikon lens. The combustion chamber was visualized from below through the fused-silica piston window using a 45° mirror, as shown in Figure 1. The field of view is 104 mm $\times$ 96 mm with a spatial resolution of 0.5 mm/pixel. Relative gain for the NL images was controlled by adjusting the exposure duration and lens aperture. In the experiments, 147 NL images were captured per fired cycle, and movies from 10 consecutive fired cycles were captured for each engine run.

Spatially integrated natural luminosity (SINL) was determined from each image by summing the intensities of all pixels, corrected for aperture setting and image exposure:

$$\text{SINL}(\text{CAD}) = \frac{f_s^2}{t_{\text{exp}} \text{SINL}_{\text{max}}} \sum_{y=1}^m \sum_{x=1}^n \text{NL}(x,y,\text{CAD})$$

In Equation 2, n and m are the number of pixels in the x - and y -directions, respectively; f_s is the lens f -stop; and t_{exp} is the image exposure duration. SINL_{max} is the signal that would be measured if all pixels within the piston bowl were saturated when the lens aperture and exposure time were set to their minimum values (i.e., $f_s = 16$ and $t_{\text{exp}} = 1.0 \mu\text{s}$). SINL values provide a time-resolved measure of high-temperature soot that can be compared between different engine runs. Further details of the NL data-acquisition procedure can be found in [15].

Chemiluminescence (CL) signal from electronically excited hydroxyl radicals (OH^*) at $308 \pm 10 \text{ nm}$ was used to determine H. The CL signal was collected every 0.5 CAD using an intensified CMOS camera (Vision Research, Phantom v311) and a Nikon 105-mm ultraviolet lens. The relative gain for each CL image was adjusted via the lens aperture and intensifier gate duration (exposure duration and intensifier gain were held constant), such that the percentage of saturated pixels fell in the range of 8-30%. Experience has shown that this range of pixel saturation provides consistent results during image processing for the measurement of H. Sequences of $114 \text{ mm} \times 114 \text{ mm}$ CL images with spatial resolution of 0.23 mm/pixel were captured. 147 CL images were captured per fired cycle, and movies from 10 consecutive fired cycles were captured for each engine run. The CL images were analyzed to determine H-values following the procedure described in detail in [15].

H measurements present inherent jet-to-jet and cycle-to-cycle variations, which are addressed by defining a mean value at each crank angle during the injection process, $\langle H(\text{CAD}) \rangle$:

$$\langle H(\text{CAD}) \rangle = \frac{1}{n_{\text{cycle}} n_{\text{jets}}} \sum_{c=1}^{n_{\text{cycle}}} \sum_{j=1}^{n_{\text{jets}}} H(j, c, \text{CAD}) \quad (3)$$

In Equation 3, $H(j, c, \text{CAD})$ represents the lift-off length measurement from a single jet and cycle, at a specific crank-angle degree; n_{jets} is the number of injector orifices, and n_{cycle} is the number of engine cycles. For notational simplicity, $\langle H(\text{CAD}) \rangle$ will be referred to as H in the rest of this paper.

The equivalence ratio at the flame lift-off length, $\phi(H)$, is estimated for each jet in each CL image using the Naber/Siebers correlation [38]. This model assumes a uniform velocity profile at the jet exit, a constant injection velocity with an instantaneous start, no velocity slip between the fuel and entrained air, and quasisteady flow with a constant spreading angle. This quantity has been shown to correlate with in-cylinder soot levels, with significant NL appearing only for conditions with $\phi(H)$ values greater than 2.

Noise and Ringing Intensity

Noise and ringing intensity were calculated per the Advanced Combustion and Emissions Control (ACEC) technical team light-duty noise guidelines [39]. Although these guidelines were written with light-duty applications in mind, useful fuel-effect comparisons can be drawn for both light-duty and heavy-duty applications from these experimental results.

As detailed in the guidelines, combustion noise is an estimate of the magnitude of audible noise or “loudness” that radiates from an engine, including frequency content. In this paper, noise is calculated using cylinder-pressure data via a post-processing script similar to that published by Shahlari et al. [40]. The algorithm involves transforming raw, single-trace, unfiltered cylinder-pressure data into the frequency domain, filtering the resulting frequency spectrum with two filters representing both engine structure attenuation and human ear frequency perception (A-weighting), and finally converting the remaining spectrum into a single value representing the loudness that would be perceived outside of an engine. Noise in absolute pressure units is calculated in this way for 180 fired cycles, and the mean noise for a given condition is then reported in units of dBA.

Ringing intensity (RI) is calculated with Equation 4 using the method of Eng [41]. Here, γ is the ratio of specific heats, β is a scale factor representing the ratio of the pressure pulsation amplitude to the maximum rate of pressure rise (in this paper this is assumed to be 0.05 ms), dP/dt_{max} is the maximum derivative of the cylinder pressure as calculated from a gradient of consecutive data points, P_{max} is the maximum cylinder pressure, R is the specific gas constant, and T_{max} is the maximum bulk- gas cylinder temperature as determined from heat release analysis.

$$\text{RI} \approx \frac{1}{2\gamma} \left(\beta \frac{dP}{dt_{\text{max}}} \right) \frac{1}{P_{\text{max}}} \sqrt{\gamma R T_{\text{max}}} \quad (4)$$

RI is measured separately from combustion noise because it measures a transient, high-intensity sound that can vary by as much as a factor of six for a given noise level [40]. The recommended maximum RI for combustion development is 5 MW/m² for all speeds and loads [39].

Engine Operating Conditions

2-Hole Tip Experiments

Table 4 presents the engine operating conditions that were held constant for the 2-hole tip experiments. The engine was fired (i.e., fuel was injected) once every 5th cycle at 1500 rpm for a minimum of 180 fired cycles per engine run for the 2-hole data. To verify data repeatability, at least three runs (i.e., 540 engine cycles) of pressure-based data were recorded at each operating condition, often on different days. As a result, data points presented in the figures throughout this paper represent averages over the three replicate measurements at the same test condition; error bars on the figures denote minimum and maximum observed values on those three separate optical-engine runs. The coefficient of variation of gross indicated mean effective pressure (gIMEP) for the 2-hole tip results varied between ~1.0% and 1.5%.

The 2-hole tip experiments were performed at three injection pressures (80, 180, and 240 MPa), with constant engine speed, intake pressure, and intake temperature, except at the highest injection pressure, where intake-manifold temperature was lowered to 50 °C in an attempt to achieve LLFC by further increasing H and decreasing $\phi(H)$. Three dilution levels-21, 18, and 16% intake- O_2 mole fraction (Y_{O_2}) were used to simulate the effects of using zero to moderate EGR rates.

Table 5 shows the mole fractions of O_2 , N_2 , CO_2 , and argon (Ar) in each intake mixture, as well as the intake-mixture densities (ρ) at -10 CAD ATDC. Because the skip-fired operating mode that was employed in these experiments affects the exhaust-gas composition, real exhaust-gas recirculation (EGR) was not used to achieve these dilution levels. Instead, X_{O_2} was varied by adding N_2 and CO_2 to the dry intake air. The experiment was designed such that the temperature, density, pressure, and specific heat at -10 CAD ATDC under motored conditions remained roughly constant for each charge-gas mixture at 760 K, 30 kg/m³, 65 bar, and 32 J/mol/K, respectively.

Table 4. Selected engine operating parameters for experiments with the 2-hole tip.

Description	Value
Engine speed	1500 rpm
Intake manifold pressure (abs.)	2.50 bar
Intake manifold temperature	95.0 °C @ 80 & 180 MPa p_{inj} 50.0 °C @ 240 MPa p_{inj}
Coolant temperature	95 °C
Fuel injection pressure	80, 180, 240 MPa
Intake oxygen	21, 18, 16% X_{O_2}
Injection duration (actual)	3500 μ s
Start of combustion	-5.0 CAD ATDC

Table 5. Species mole fractions (X) and densities (ρ) of the studied charge-gas mixtures for the 2-hole tip experiments.

Parameter	21% X_{O_2}	18% X_{O_2}	16% X_{O_2}
X_{O_2} [mol%]	20.946	18	16
X_{N_2} [mol%]	78.080	80.899	82.811
X_{CO_2} [mol%]	0.040	0.298	0.476
X_{Ar} [mol%]	0.934	0.803	0.713
ρ at -10 CAD ATDC [kg/m ³]	29.72	29.62	29.56

Thermodynamic conditions were matched at -10 CAD ATDC rather than at TDC for three main reasons. First, it was desired to keep the thermodynamic conditions at the start of fuel injection the same independent of fuel type or operating condition. Hence, thermodynamic conditions were matched at -10 CAD ATDC, because this is approximately where fuel injection begins. Second, the start of combustion (SOC) was held constant at -5 CAD ATDC for all operating conditions (see Table 4), so the thermodynamic conditions at TDC vary considerably with injection pressure, load, dilution level, and ignition processes, making it less meaningful to match thermodynamic conditions at TDC. Third, due to the optical engine having a lower compression ratio than its corresponding metal engine, its pressure and temperature do not vary as rapidly with crank angle as in the metal engine (see Appendix A of [27]). Hence, matching thermodynamic conditions at -10 CAD ATDC provides a better match of conditions throughout the period of fuel vaporization and combustion between optical and metal engines than does matching thermodynamic conditions at TDC. The constant specific-heat value was used to isolate the effects of X_{O_2} variation on the results (i.e., independent of the specific heat variation) and to reduce the rate of consumption of bottled CO_2 during experiments (real EGR contains higher CO_2 levels).

Injection timing and duration were chosen to obtain predominantly mixing-controlled combustion and valid H measurements. As indicated in Table 4, the SOC timing was held constant for all fuels

and operating conditions, which required the injection timing to be varied with fuel type and operating conditions. Under some of the conditions investigated herein, a measurable amount of low-temperature heat-release (LTHR) was produced, such that the zero crossing of the apparent heat-release rate (AHRR) would often measure the start of LTHR rather than the start of high temperature heat release. This is addressed by using a parameter called SOC_{fit} rather than SOC, which is defined as the zero crossing of a line that intersects the points where the AHRR trace first attains 20% and 60% of its peak value (see Figure 4 of [15]). Table 6 shows that the similar injection duration for T50 and CFA operation resulted in a 25% load difference due to the lower q_{LHV} for T50 relative to that for CFA.

Table 6. Engine load variations (gIMEP [bar]) with fuel and injection pressure for the 2-hole tip experiments. Injection duration was held constant for these experiments, which led to load variations with injection pressure and fuel type.

Injection Pressure	CFA	T50
80 MPa	1.72±0.03	1.25±0.03
180 MPa	2.91±0.03	2.19±0.03
240 MPa	3.12±0.03	2.64±0.02

A portion of the 2-hole tip experimental data has been published previously by Dumitrescu et al., including CFA and T50 data at 80 and 180 MPa injection pressures at all dilutions [22], and T50 data at 240 MPa injection pressure for all dilutions [42], all at 95 °C IMT. These data are shown for reference in this paper alongside newly acquired data for CFA at 240 MPa and 50 °C IMT in Figures 6, 7, 8, 9, 10.

6-Hole Tip Experiments

Table 7 presents the engine operating conditions for the 6-hole tip experiments. The engine was fired once every 15th cycle (compared to once every 5 cycles with the 2-hole tip) for a minimum of 60 fired cycles per engine run. To verify data repeatability, at least three engine runs of pressure-based data were recorded at each operating condition, sometimes on different days. For the 6-hole tip, the coefficient of variation of gIMEP was always very low: between 0.4-0.7%. Similar to the 2-hole tip results, data points presented in the figures represent averages over the three replicate measurements at the same test conditions, and error bars represent the minimum and maximum observed values on three separate optical-engine runs.

Table 7. Selected engine operating parameters for experiments with the 6-hole injector tip.

Description	Value
Engine speed	1500 rpm
Intake manifold pressure (abs.)	2.50 bar
Intake manifold temperature	30.0 °C
Coolant temperature	85 °C
Fuel injection pressure	240 MPa
Intake oxygen	21, 16% X_{O_2}
Injection duration (actual)	2410 μ s (for T50), 2020-2070 μ s (CFA) ^a
Start of combustion	-5.0, 0.0, +5.0 CAD ATDC

^aThe gIMEP between T50 and CFA was matched, so the injection duration for CFA varied.

The experiments were performed at a fixed injection pressure (240 MPa), engine speed (1500 rpm), intake pressure (2.50 bar), and intake temperature (30 °C). Two dilution levels 21 and 16% X_{O_2} were used to simulate the effects of using zero to moderate